MULTI-JOURNAL BEARING MOTOR

CROSS-REFERENCE TO RELATED APPLICATIONS

[0001] This new application for letters patent claims priority from an earlier filed provisional patent application entitled "Multi-Journal Bearing Motor." That application was filed on April 18, 2003 and was assigned Application No. 60/464,001.

BACKGROUND

Field of Invention

[0002] The present invention relates to fluid dynamic bearing motors and, more specifically, to a multi-journal fluid dynamic bearing motor assembly.

Description of Related Art

[0003] **Figure 1** provides a perspective view of a disc drive assembly **150**. In this arrangement, a plurality of discs **110'** are stacked vertically within the assembly **150**, permitting additional data to be stored, read and written. The drive spindle **151** receives the central openings **105** of the respective discs **110**. Separate suspension arms **156** and corresponding magnetic head assemblies **158** reside above each of the discs **110**. The assembly **150** includes a cover **130** and an intermediate seal **132** for providing an air-tight system. The seal **132** and cover **130** are shown exploded away from the disc stack **110'** for clarity.

[0004] In operation, the discs **110** are rotated at high speeds about the spindle **151**. As the discs **110** rotate, an air bearing slider on the head **158** causes each magnetic head **158** to be suspended relative to the rotating disc **110**. The flying height of the magnetic head assembly **158** above the disc **110** is a function of the speed of rotation of the disc **110**, the aerodynamic lift properties of the slider along the magnetic head assembly **158** and, in some arrangements, a biasing spring tension in the suspension arm **156**.

[0005] A servo spindle **152** pivots about pivot axis **140**. As the servo spindle **152** pivots, the magnetic head assembly **158** mounted at the tip of its suspension arm **156** swings through arc **142**. This pivoting motion allows the magnetic head **158** to change track positions on the disc **110**. The ability of the magnetic head **158** to move along the surface of the disc **110** allows it to read data residing in tracks along the magnetic layer of the disc. Each read/write head **158** generates or senses electromagnetic fields or magnetic encodings in the tracks of the magnetic disc as areas of magnetic flux. The presence or absence of flux reversals in the electromagnetic fields represents the data stored on the disc.

[0006] Fluid dynamic bearings tend to generate less vibration and non-repetitive run-out in the rotating parts of motors than ball bearings and other types of bearings. For this reason, fluid dynamic bearing motors are oftentimes used in precision-oriented electronic devices to achieve better performance. For example, using a fluid dynamic bearing motor in a magnetic disc drive, such as magnetic disc drive **150** described above in conjunction with **Figure 1**, results in more precise alignment between the tracks of the discs and the read/write heads. More precise alignment, in turn, allows discs to be designed with greater track densities, thereby allowing smaller discs and/or increasing the storage capacity of the discs.

[0007] As persons skilled in the art are aware, an ongoing challenge in fluid dynamic journal bearings is balancing the tradeoff between motor performance and power consumption. For example, increasing the stiffness of the fluid dynamic journal bearings results in less vibration in a motor's rotating parts and, therefore, increased motor precision and performance. However, an increase in the stiffness of the bearings is usually accompanied by an increase in the power consumption of the motor. Therefore, there exists a need for a technique to increase the stability of a fluid dynamic bearing without increasing the amount of power consumed by the fluid dynamic bearing.

SUMMARY OF THE INVENTION

[0008] One embodiment of a method for designing a fluid dynamic bearing system includes determining a first stability ratio for a first journal bearing configuration. The method further includes determining a second stability ratio for a second journal bearing configuration. The two stability ratios may then be compared. Preferably, the first configuration has two sub-journal bearings and the second configuration has three sub-journal bearings. A third stability ratio for a third configuration may then be determined if the second stability ratio is greater than the first stability ratio. The third configuration may have four sub-journals.

[0009] The disclosed method is especially useful for designing fluid dynamic bearing systems. One advantage of the disclosed method is that a journal arrangement designed according to the disclosed method has substantially greater stability than a journal arrangement not designed according to the disclosed method. Further, neither the radial stiffness nor the power consumption of the journal arrangement designed according to the disclosed method decreases appreciably.

BRIEF DESCRIPTION OF THE DRAWINGS

[0010] So that the manner in which the above recited features of the present invention can be understood in detail, a more particular description of the invention, briefly summarized above, may be had by reference to the appended drawings. It is to be noted, however, that the appended drawings illustrate only typical embodiments of this invention and are therefore not to be considered limiting of its scope.

[0011] Figure 1 illustrates a perspective view of an exemplary disc drive assembly as might employ the improved spindle motor arrangement of the present invention,

[0012] Figure 2 illustrates a cross-sectional view of an improved spindle motor, according to one embodiment of the invention, in which three distinct journal bearings are provided,

[0013] Figure 3 illustrates the effects of the radial stiffness and the cross-coupled stiffness of a typical fluid dynamic journal bearing on the motion of a shaft, according to one embodiment of the invention,

[0014] Figure 4 illustrates the results of a simulation comparing the vibration responses of motors employing two, three, and four journal bearings, according to one embodiment of the invention, and

[0015] Figure 5 is a flow chart of method steps for using the stability ratio in the design of a fluid dynamic bearing, according to one embodiment of the present invention.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENT

[0016] **Figure 2** presents a partial cross-sectional view of an improved spindle motor arrangement **200** in one embodiment, in which three journal bearings **275**, **280**, and **285** are provided. The motor **200** first comprises a hub **210**. The hub **210** includes an outer radial shoulder **205** for receiving a disc (not shown in **Figure 2**). Disposed within the hub **210** is a sleeve **250**. During operation, the sleeve **250** and hub **210** rotate together. In this arrangement, the sleeve **250** resides and rotates on a non-rotational thrust washer **265**. A shaft **255**, which is coupled to a base **230**, is provided along the inner diameter of the sleeve **250** to provide lateral support to the sleeve **250** while it is rotated. Joined to a bottom surface of the sleeve **250** is a shield **260**. The shield **260** encloses an outer portion of the thrust washer **265**. Between an inner side of the shield **260** and the enclosed portion of the thrust washer, a gap **295** is formed. The motor **200** also includes a stator **245**, which is mounted on a base **230**. The stator **245** typically defines an electric coil that, when energized, creates a magnetic field. The energized coil cooperates with magnets **215**, which are mounted from an inner surface of the hub **210** on a back iron **240**, to cause the hub **210** to rotate relative to the shaft.

[0017] As persons skilled in the art will recognize, the motor **200** includes a hydrodynamic bearing system. More specifically, the thrust washer **265** is disposed proximally to a bottom surface of the sleeve **250**, and fluid is injected in gaps

maintained between the sleeve **250** and surrounding parts, e.g., the shaft **255** and the thrust washer **265**, through a fill hole **270** disposed through the shield **260**. The fluid defines a thin fluid film that supports relative movement of the parts. The interface between the bottom of the sleeve **250** and the top of the thrust washer **265** thus defines a thrust bearing **290**. Liquid lubricant is provided along the thrust bearing gap **291** to provide a fluid bearing surface. Either a top face of the thrust washer **265** or a bottom surface of the sleeve **250** may include a grooved pattern (not shown) for receiving and holding liquid lubricant when the motor **200** is at rest. When the motor **200** is at rest, the sleeve **250** presses directly on the thrust washer **265**. Fluid is then extruded around the outer diameter of the shaft **155** and into a shaft-sleeve gap **292** and/or a shield-thrust washer gap **295**.

[0018] When the motor **200** is energized and the sleeve **250** and adjoining hub **210** are rotated, lubricating fluid is drawn into the thrust bearing region **290** to support relative rotation between the bottom end of the sleeve **250** and the facing surface of the thrust washer **265**. To limit the axial displacement of the sleeve **250** and adjoining hub **210** during operation, an axial bias force is typically introduced. In this embodiment, a bias ring **235** is utilized to prevent the thrust bearing gap **290** from becoming too large and reducing the stiffness of the thrust bearing. In such a configuration, an axially downward magnetic force results that pulls magnet **215** (and therefore hub **210**) towards base **230**. The magnitude of this force is a function of, among other things, the size of a gap between the magnet **215** and the bias ring **235**. Alternatively, base **230** may comprise a magnetic metal such as a Series 400 steel or a low carbon steel. In other alternative embodiments, the biasing force may be created in any other feasible way such as, for example, by applying a spring force or a downward-acting pressure force on hub **210**. When the rotating sleeve **250** comes to rest, the sleeve end will rest on the thrust washer **265**. Although the volume of fluid is very small, it will tend to be forced back out into the gap(s) **295** and/or **292**. Therefore, space is preferably allowed in this gap(s) **295** and/or **292** for this fluid. To inhibit the loss of liquid lubricant from the gaps **290** and **292**, optional capillary seals are provided along gaps **295** and **294**, respectively. When the motor

200 is idle, the capillary seals **295**, **294** aid in maintaining fluid within the bearing system.

[0019] In addition to thrust bearing **290**, **Figure 2** shows that there are three sets of groves disposed on an outer surface of the shaft **255** along a shaft-sleeve interface **292**. The grooved portions of the shaft along with the corresponding outer sections of the sleeve **250** comprise top **275**, intermediate **280**, and bottom **285** journal bearings, respectively. Upon rotation of the sleeve **250**, the grooved patterns of each of the journal bearings **275**, **280**, **285** create a high pressure region at the center of each pattern to retain fluid and provide stiffness to the motor **200**. Although, only one intermediate journal **285** is depicted in **Figure 2**, any number of intermediate journals **285** may be provided, depending upon design considerations of a particular motor.

[0020] The substantially chevron-shaped groove patterns of the journals **275**, **280**, and **285** shown in **Figure 2** are preferred but other patterns, such as spiral patterns, would suffice. Preferably, the groove patterns are disposed on the shaft **255**, however, they may also be disposed on an inner surface of the sleeve **250**. It can seen that the groove patterns are each identical to one another and spaced equidistantly along the shaft **255**. This is only a preferred embodiment, however. In alternative embodiments, the journal bearings **275**, **280**, and **285** may each comprise a different number and configuration of grooves and may be unequally spaced along the gap **292**. Further, the grove patterns may overlap one another as long as they are substantially separate, i.e., as long as the apex(es) of each groove pattern is/are longitudinally spaced apart.

[0021] It is understood that the motor seen in **Figure 2** is exemplary only. The present invention may be employed in various motor configurations. For example, the motor may be configured so that the shaft is coupled to the hub and rotates around a stationary sleeve.

[0022] **Figure 3** illustrates the effects of the radial stiffness **320** and the cross-couple stiffness **325** of a typical fluid dynamic journal bearing **330** on the motion of a

shaft **305**, according to one embodiment of the invention. As shown, the shaft **305** is subject to an excitation force that causes the shaft **305** to move in the horizontal direction towards a sleeve **310**. As a result of this horizontal motion **315**, the fluid dynamic journal bearing **330** exerts a force on the shaft **305** in a direction parallel and opposite to the horizontal motion **315**. The radial stiffness **320** (k_{xx}) of the fluid dynamic journal bearing **330** causes this parallel and opposite reaction force. This type of stiffness is a desirable quality in fluid dynamic journal bearing **330** because the radial stiffness **320** tends to reduce the amplitude of the horizontal motion **315**. In addition to the radial stiffness **320**, the fluid dynamic journal bearing **330** is configured to have a cross-coupled stiffness **325** (k_{xy}), which acts orthogonally to the radial stiffness **320**. As persons skilled in the art will understand, the cross-coupled stiffness **325** causes the fluid dynamic journal bearing **330** to exert a force on the shaft **305** in a direction orthogonal to the horizontal motion **315**. This orthogonal force, in turn, introduces an orthogonal component **316** to the motion of the shaft **305**, thereby decreasing the stability and performance of fluid dynamic journal bearing **330**. For this reason, cross-coupled stiffness **325** is not a desirable quality in fluid dynamic journal bearing **330**.

[0023] The stability of a bearing may be gauged by the radial stiffness divided by the cross coupled stiffness (stability ratio). A high ratio indicates a relatively greater radial stiffness, meaning that the radial stiffness governs the behavior of the shaft more than the cross-coupled stiffness. Thus, with a greater ratio, the propensity of a fluid dynamic journal bearing, through the radial stiffness, to limit the shaft motion outweighs the propensity of the journal, through the cross-coupled stiffness, to add unwanted motion to the shaft.

[0024] As persons skilled in the art are aware, shorter journal bearings tend to have greater stability ratios than longer ones. However, simply reducing the length of the journal bearings in a conventional two-journal motor to improve stability would have negative consequences, such as a loss of radial stiffness. In order to obtain the stability benefits of a shorter bearing while not sacrificing the radial stiffness of a longer bearing, the overall journal length of the two-journal motor may be broken into a greater number of sub-journals. Doing this, effectively takes a larger journal with a

lower stability ratio and divides it into smaller journals with greater stability ratios. Because these smaller sub-journals act in parallel, the overall effective journal length remains approximately constant, so the radial stiffness remains relatively constant. Further, because the radial stiffness doesn't change, power consumption remains constant (or, more importantly, does not increase). Because smaller-journals with greater stability ratios are used, the effective stability ratio for the overall journal increases, thereby decreasing or masking the influence that the cross-coupled stiffness has on shaft motion.

[0025] **Figure 4** shows the results of a simulation measuring the response of a fluid dynamic bearing motor with two **400**, three **410**, and four **420** journal configurations at different operating frequencies, according to one embodiment of the invention. More specifically, the performance of each motor configuration is shown as the operational vibration (op-vibe) of the motor as a function of frequency. The journal gap of each motor configuration is the same. The total journal bearing length of each motor is the same, therefore, power consumption is approximately the same. In each of the three simulations, the total bearing length was divided into two **400**, three **410**, and four **420** "sub-journals," respectively. As **Figure 4** shows, for frequencies between about 200 and about 600 Hertz, the four-journal configuration **420** exhibits stability superior to either the two **400** or three **410** journal configurations. Further, for frequencies between 400 and about 800 Hertz, the three journal configuration **410** exhibits superior stability to the two journal **400** configuration. The foregoing shows that in certain circumstances, dividing up the total journal length into smaller sub-journals may increase performance without increasing the power consumption of the motor.

[0026] **Figure 5** is a flow chart of method steps **500** for using the stability ratio in the design of a fluid dynamic bearing, according to one embodiment of the present invention. Although the method steps **500** are described in the context of the systems illustrated in **Figure 1-4**, any system configured to perform the method steps in any order is within the scope of the invention.

[0027] As shown in **Figure 5**, the method of using the stability ratio starts in step **510** where fluid dynamic journal bearing parameters are determined according to total system requirements, such as, for example, power consumption, total radial stiffness, and loading. These requirements are usually supplied by a customer. Typically, these requirements fix bearing parameters, such as the shaft diameter, the journal gap between the shaft and the sleeve, and the total length of the journal bearings.

[0028] At step **520**, the motor is initially designed with a conventional two journal bearing configuration. At step **530**, the stability ratio of the two journal motor is determined. The stability ratio may be determined theoretically or empirically. At step **540**, an additional journal bearing is added to the initial two-journal design. Preferably, this is done by taking the total journal bearing length of the two journal design and providing three sub-journals, each one third the length of the total journal bearing length.

[0029] At step **550**, the stability ratio of the motor with the three journal configuration is determined. As previously discussed herein, decreasing the individual sub-journal bearing lengths may improve the stability ratio of a given motor. There are situations, however, in which this will not be the case. For example, referring back to **Figure 4**, it can be observed that for frequencies less than about 200 Hertz, the stability ratio of the two journal configuration is optimal and for frequencies greater than about 600 Hertz and less than about 800 Hertz, the stability ratio of the three journal configuration is optimal. Thus, at step **560**, the stability ratio of the three journal design is compared to that of the conventional two journal design to see if the stability ratio has increased by adding another sub-journal. If so, then the method returns to step **540**, and the process is repeated until adding an additional sub-journal no longer increases the stability ratio of the motor. At that point, the stability ratio is optimized and the design is concluded. If the stability ratio of the three journal design is not greater than that of the two journal design, then the two journal configuration may be deemed optimal and the design is concluded.

[0030] The invention has been described above with reference to specific embodiments. Persons skilled in the art, however, will understand that various modifications and changes may be made thereto without departing from the broader spirit and scope of the invention as set forth in the amended claims.